A Tool For Design Minimization Of Aircraft Interior Noise

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Abstract

The design and development of new generations of aircraft of all classes continues apace. Increasingly, the quality of the interior acoustic environment of such new aircraft is becoming of greater importance. A need exists, then, to address the integration of acoustic considerations into the aircraft design process. To address this need, a computational design tool was developed to perform a constrained optimization of the acoustic environment within a vibrating cylinder, incorporating finite element and boundary element methods. The tool comprises a UNIX shell script that coordinates an iterative design optimization process integrating a number of programs, the key components of which are: NASTRAN for structural analyses; COMET/Acoustics for acoustic analyses; and CONMIN for nonlinear optimization. In addition to the structure and implementation of the tool, this paper presents the results of a number of trials of the tool applied to unstiffened cylinders, considering different formulations of the objective function to be optimized. The results presented here are principally directed toward tonal noise components within turbo-prop driven aircraft.

Glossary

x	Design variable vector
x_L, x_U	Vectors of lower and upper bounds
$f(x)$	Objective function
f_o	Initial value of objective function
W	Structural weight
W_{Max}	Constrained maximum weight
W_o	Initial structural weight
W_{Middle}	Weight of the middle design point model
$\hat{p}_i(\mathbf{x})$	Complex acoustic pressure at node i
β_a	Auxiliary acoustic design variable
β_w	Auxiliary weight design variable
μ_a	Auxiliary acoustic design variable weighting
μ_w	Auxiliary weight design variable weighting

Introduction

We present here the end results of four different objective and constraint formulations, implemented within a computational design tool, for the optimization of unstiffened cylinder models. The desired goals of the optimizations are related to the structural weight and interior noise environment of the cylinder models. The cylinders are considered to be excited at a single frequency by an external noise source, modeled here as a monopole. For this unstiffened model, the cylinder wall thicknesses are considered to be the design variables. The cylinders considered here are not representative of aircraft structures, per se, but are suitable for assessing trends in optimized cylinders considering both weight and interior noise criteria, which is the key focus of this paper.

There has recently been increasing interest among airlines and aircraft manufacturers in controlling the acoustic environment within aircraft cabins in order to provide passengers with a more comfortable environment, and to provide the flight crew with a flight deck environment that does not impair communication. In turboprop aircraft, the interior noise is dominated by noise due to the propeller blades. A propeller produces a highly tonal and highly directional sound field that is dependent on blade thickness, steady thrust, and blade tip speed, and is therefore deterministic. No current certification requirements exist to regulate interior noise, but airlines do require noise level guarantees of the manufacturer.[1] Airframers have confronted this problem by two broad means: active and passive noise control.[2, 3] Active noise control essentially involves the placement of loudspeakers and microphones throughout the passenger cabin or flight deck in conjunction with a control system. Active noise control is an appealing technique since it can be fitted to existing airframes without major structural modification. Passive noise control, by contrast, seeks to reduce cabin noise by the modification of the airframe structure itself. For example, this could be in the form of acoustic treatments added to the cabin walls.

It is desirable that future aircraft be designed to have quieter passenger cabins, minimizing the need for supplementary active or passive noise control systems. Designing quiet structures is itself a form of passive noise control, and falls under the broad category of Multidisciplinary Design Optimization (MDO).[4] The objective, constraint, and goal functions in an MDO problem model the often conflicting goals of a variety of disciplines. Design of quiet aircraft cabins is multidisciplinary in that it requires satisfaction of not only acoustic goals and constraints, but also structural goals and constraints such as weight, stress and aeroelastic effects. This makes the multidisciplinary design process extremely complicated, requiring many iterations of time consuming analyses coupled with a mathematical programming algorithm as a means of optimizing the system. In addition, due to the complexity and often high nonlinearity of MDO problems, it is nearly impossible to guarantee that a solution is truly optimal. It remains the task of the designer to choose from among a set of candidate solutions the one which best satisfies not only the goals and constraints defined in the MDO problem, but also meets more intangible standards such as design experience and intuition. Although MDO may be difficult to implement, it offers a number of far reaching benefits to the designer: (1) goal-orientation of the design approach; (2) tedium reduction; (3) decision-making help, trend finding, and preliminary design guidance.[5]

Some examples of the use of structural acoustic optimization in industry have appeared in the literature.[6-8] The methods used were all computational, since automation of design optimization was necessary to keep design times down. In the present work, an approach similar to that used in earlier research is used to optimize the design of aircraft fuselage structures in order to minimize the noise in the interior. This algorithm is implemented through a computational design tool.[9] This form of passive noise control involves coupling structural and acoustic analysis packages with an optimization algorithm as part of an automated process. The heart of the design tool is a UNIX shell script that controls the automated iterative analysis and optimization procedure. The user provides a group of input data files describing the structural and acoustic models and the parameters of the optimization procedure, and the design tool returns updated data files describing the optimized sound field and design of the structural model. The purpose of the overall research project is to develop and validate a flexible, robust tool for the optimal design of stiffened or unstiffened cylinders subject to various acoustic and structural objectives and constraints and excited by an external harmonic noise source.

The following section introduces the computational structure of the design tool, and describes the specific models to which the optimization formulations will be applied. Next, the four optimization formulations considered in this paper are introduced. The subsequent section presents the results pertaining to the formulations' performance, followed by a section with our conclusions.

Description of the Design Tool

The design tool is described in greater detail in earlier papers[9, 10]. We will briefly review its features and capabilities here. The design tool is a UNIX shell script that controls the interaction of several main programs and supporting programs and implements error-trapping capabilities. Figure 1 is a flowchart depicting the structure of the script's algorithm.

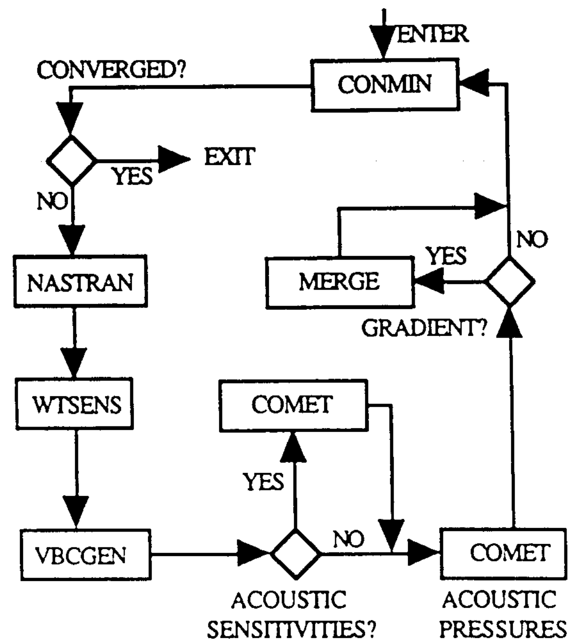


Figure 1 - Schematic of design tool algorithm logic

The main programs within the design tool are CONMIN, NASTRAN, and COMET/Acoustics. CONMIN[11] is the actual optimizer. It uses the Modified Method of Feasible Directions algorithm to optimize a single objective function subject to inequality constraints and side constraints on the design variables.[12] Structural analyses are performed with MSC/NASTRAN Version 68. All structural models are assumed to have structural damping. The models are presumed to be excited by external acoustics loads. The external acoustic loads are computed in advance using the boundary element code COMET/Acoustics, with the pressures applied as harmonically varying loads at the structural element nodes. COMET/Acoustics is used within the design tool to perform the acoustic analyses of the cylinder interior.[13] At present, the acoustic analysis for the cylinder models is uncoupled, meaning that the vibration of the structure is not considered to be affected by the bounding acoustic medium.

WTSENS, VBCGEN and MERGE are supporting codes for the main programs. WTSENS extracts the total cylinder weight and weight sensitivities from NASTRAN output. The weight sensitivities are the change in weight due to a change in a design variable (in the context of this paper, the design variables are shell element thicknesses). VBCGEN translates surface velocities from NASTRAN into a COMET/Acoustics data set as velocity boundary conditions.

MERGE combines structural design sensitivities from NASTRAN (change in velocity due to change in design variable) and acoustic velocity sensitivities from COMET (changes in acoustic pressures due to changes in surface normal velocity) to produce the acoustic design sensitivity (change in acoustic pressure due to change in design variable). The acoustic sensitivities from COMET/Acoustics need be computed only once for each cylinder model, at the start of the optimization procedure. If the same model is optimized later (but at the same frequency), perhaps with a different objective or constraints, the same sensitivity data files are reused.

The user must prepare several files before invoking the script: A file containing parameters that define convergence criteria and constraint tolerances for CONMIN, desired objective formulation, a NASTRAN dataset describing the structural model and the external loading conditions, and input files for COMET/Acoustics. The COMET/Acoustics input files include such information as the interface between the structural and acoustic models, and the locations of points within the model at which to compute acoustic pressures.

Cylinder Models

We consider in this work unstiffened cylinders. The unstiffened cylinder is considered to be made of aluminum, 3.66 m in length with a radius of 0.838 m. The shell thickness upper and lower bounds are 1.0 mm and 3.4 mm, respectively, and the shell has an initial thickness of 1.7 mm. Both ends of the cylinder are clamped. The physical dimensions of the cylinder model are the same as the outer shell of the model used by Grosveld et al. in their study of active structural acoustic noise control.[2]

Since only symmetric modal response is being investigated at this time, analysis is simplified by modeling only one quarter of the cylinder. The base NASTRAN model for the unstiffened cylinder is composed of 800 linear quadrilateral elements. To study the influence of longitudinal variations in the design variables, we grouped the shell elements into 20 circumferential bands of 40 elements each. We refer to this model as the circumferential model. To study the influence of circumferential variations in the design variables, we grouped the shell elements into 20 longitudinal bands of 40 elements each. We refer to this model as the longitudinal model. Each of these models has 20 design variables, representing the thickness of the elements in each of the 20 groups. The design variable groupings and their numerical designations are depicted in Figs. 2 and 3.

The design variables for the shell groups are shell thicknesses. Bounds on the shell thickness are set at 1.0 mm and 3.4 mm, with a reference thickness of 1.7 mm. We assume the cylinder is excited by a single exterior monopole source at a frequency of 154.2 Hz. This forcing excites a (2,1) structural resonance in the cylinder with design variables at reference conditions. The source is located 0.168 m to one side of the cylinder, halfway down its length. The acoustic loads produced by this source on the cylinder are computed beforehand by COMET and included in the NASTRAN dataset. We consider two additional models with different starting design states. These other models are initially not on resonance. Further, in all instances we determine the effectiveness of the optimization based on the ending noise level, not on the reduction from the starting point. Therefore, the fact that the reference models start on resonance is not significant.

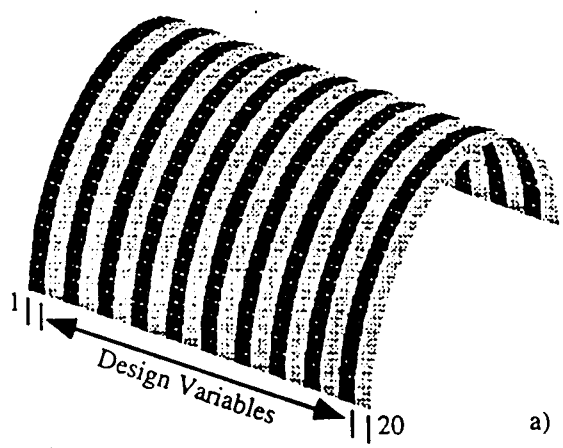


Figure 2 - a) Unstiffened circumferential model.

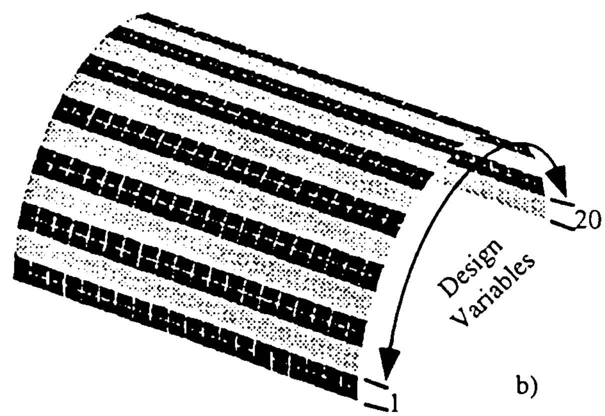


Figure 3 - b) unstiffened longitudinal model.

Optimization Problem Formulations

The following briefly presents the mathematical formulation for each of the objective function/constraint options implemented in the design tool. An assessment of these formulations' relative performance may be found in our earlier paper.[10]

"Baseline" Formulation

The "Baseline" formulation minimizes the sum of the squares of the acoustic pressure amplitudes at discrete data recovery points within the volume, while side constraints bound the design variables:

$$\text{Minimize} \quad f(x) = \frac{\sum_{i=1}^{NDRN} \tilde{p}_i(x)\, \tilde{p}_i^*(x)}{f_0} \tag{2}$$

$$\text{Subject to} \quad x_L \le x \le x_U . \tag{3}$$

In Eq. (2), $\tilde{p}_i(x)$ is the acoustic pressure at the ith data recovery node, the asterisk indicates complex conjugation, and NDRN is the number of data recovery nodes within the volume of interest. The objective function is scaled by f_0, the value of the objective at the initial design state. Lower and upper bounds on the design variables are represented by x_L and x_U, respectively.

"Acoustic" Formulation

The "Acoustic" formulation minimizes the sum of the squared acoustic pressures subject to design variable bounds and to a constraint on the total weight:

$$\text{Minimize} \quad f(x) = \frac{\sum_{i=1}^{NDRN} \tilde{p}_i(x)\, \tilde{p}_i^*(x)}{f_0} \tag{4}$$

$$\text{Subject to} \quad \frac{W}{W_{MAX}} - 1 \le 0 \tag{5}$$

$$x_L \le x \le x_U . \tag{6}$$

In Eq. (4), f_0 is the initial value of the objective function. In Eq. (5), W is the total weight of the structure, and W_{Max} is a user-specified maximum weight.

"Weight" Formulation

The "Weight" formulation minimizes the weight of the structure subject to design variable bounds and a constraint on the maximum value of the sum of the squared pressures:

$$\text{Minimize} \quad f(x) = \frac{W}{W_0} \tag{7}$$

$$\text{Subject to} \quad \frac{\sum_{i=1}^{NDRN} \tilde{p}_i(x)\, \tilde{p}_i^*(x)}{\left(\sum |\tilde{p}|^2\right)_{MAX}} - 1 \le 0 \tag{8}$$

$$x_L \le x \le x_U . \tag{9}$$

In Eq. (7), W_0 is the initial value of the weight, while in Eq. (8) $\left(\sum |\tilde{p}|^2\right)_{MAX}$ is the maximum sum of the squared pressures. In the implementation of Eq. (8), the user specifies a desired decibel level for this maximum, and the algorithm converts it to an appropriate sum of squared pressure magnitudes.

"Compound" Formulation

The "Compound" formulation simultaneously minimizes the sum of the squared acoustic pressures and the structural weight. The objectives, the pressure sum and the weight, are normalized by their initial values and required to be less than bounds β_a and β_w, respectively. These bounds are considered to be two additional design variables, and their weighted sum is taken as the new objective function, where μ_a and μ_w are the respective weighting factors. The weighting factors allow one or the other objective to be considered more important. The mathematical statement of the formulation is:

$$\text{Minimize} \quad \mu_a\beta_a + \mu_w\beta_w \tag{10}$$

$$\text{Subject to} \quad \frac{\sum_{i=1}^{NDRN} \tilde{p}_i(x)\, \tilde{p}_i^*(x)}{\left(\sum |\tilde{p}|^2\right)_o} - \beta_a \le 0 \tag{11}$$

$$\frac{W}{W_o} - \beta_w \le 0 \tag{12}$$

$$x_L \le x \le x_U , \tag{13}$$

where $\left(\sum |\tilde{p}|^2\right)_o$ is the initial value of the sum of the squared pressures (before the optimization process begins), and similarly W_o is the initial weight of the structure.

Results

As noted above, our earlier paper addressed the relative performance of each of the objective function formulations, with respect to computational efficiency and stability. This earlier work focused on a simpler evaluation model than what we consider here. For the purposes of the present work, we focus on the optimal design states and objective functions for the more complex models.

Multiple initial designs are needed since there is no guarantee that a single global optimal solution exists for each set of objectives and constraints. Indeed, the existence of multiple local minima, each representing a local "optimum," is a distinct probability. In practice, one chooses a "suitable" number of starting points so as to "adequately" sample the design space and then examines the results for each to select the "best" optimum. For the work presented here, we chose to use three starting designs for the unstiffened model. The three states correspond to all design variables at their upper bounds, all design variables at their lower bounds, and all design variables at an approximate middle point between the bounds. Table 1 summarizse these three initial design states. In the remainder of this paper, the starting design states will be referred to as the "Upper," "Middle," and "Lower" cases, corresponding to all design variables at the upper bound, intermediate state, and lower bound, respectively. In addition, the data presented in the following sections has been normalized with respect to the design states and weights for the "Middle" starting point. Each objective function was applied to each of the three initial design states, for each of the different design variable groupings.

Table 1 - Initial designs, unstiffened model

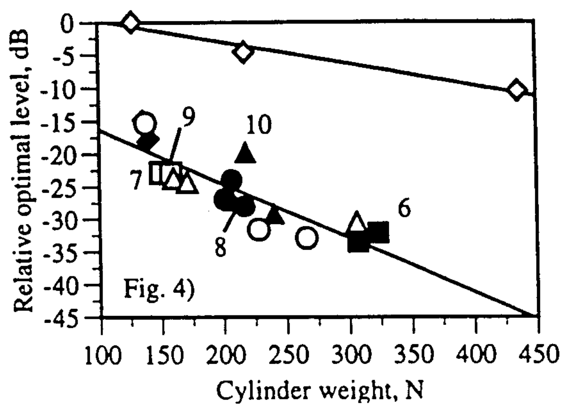
Initial State	Shell Thick [mm]	Cyl. Wt. [N]	SPL [dB]
Upper	3.4	436	-10.6
Middle	1.7	218	-4.6
Lower	1.0	128	0.0

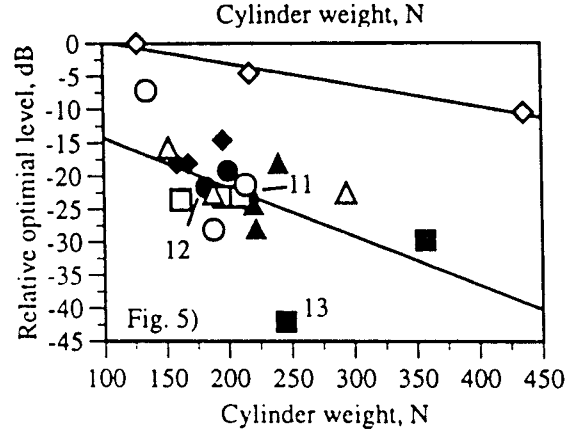
All of the models were subject to the same exterior forcing, at the same frequency. Given that the exterior monopole's excitation source strength was arbitrary, the absolute interior levels are not necessarily significant. For that reason, a common zero dB reference was chosen. The levels in Table 1 reflect this reference. All results are normalized with respect to that zero dB reference. We use the relative average sound pressure level (SPL) within the fuselage as the figure of merit in determining the impact of the optimization process. The lower this level, the quieter the interior. We also determine the relative interior level prior to optimization, and use the difference between the optimized and un-optimized levels as a further indicator of the impact of the optimized design.

The following results for the unstiffened model are a subset of the number of analyses we have performed. They illustrate the variations in the ending design states for different starting states, as well as the variations in the optimal design states generated through the different objective functions.

Unstiffened model results

Figure 4 presents the relative optimal SPL levels obtained with the circumferential model, and similarly with Fig. 5 for the longitudinal model, for the various cases analyzed here. Note that the numbers next to individual data points in these figures correspond to later figure numbers for the corresponding design state. Figures 4 and 5 also include data for uniform thickness cylinder models of different weights, as a means of providing a reference between optimized and unoptimized cylinders. Linear curve fits have been applied to the data, yielding the lines on the figures. The upper line in both figures is for the data from the unoptimized models, and since all of the data from the optimized models is below this line, the optimization indicates that non-uniform shell thickness can provide reduced levels of interior noise. Another observation that may be made is that, within each objective function and constraint combination, there is no single global optimum. Indeed, all 20 of the analyses represented in Fig. 4 yielded different ending design states, as did the 20 analyses in Fig. 5.

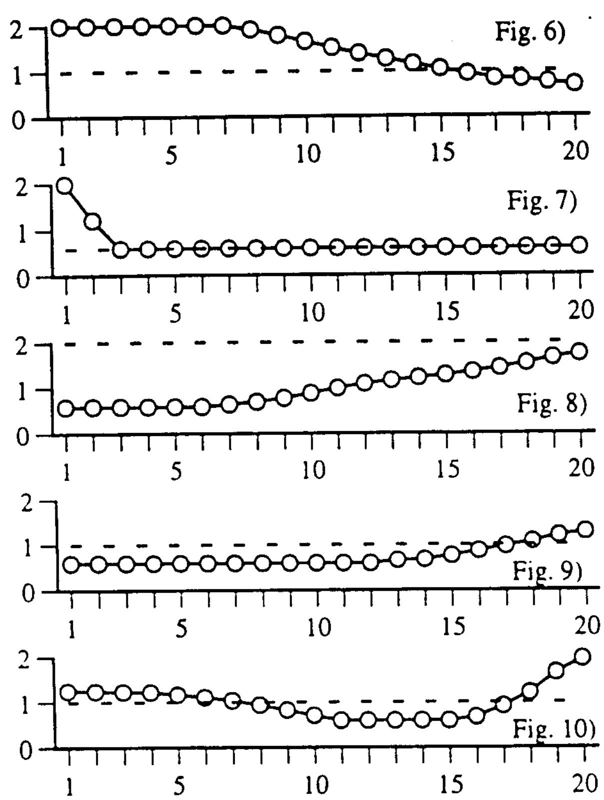




Figures 4 - 5: Optimal relative SPL vs. model weight for unstiffened circumferential 4) and longitudinal 5) models. ■ - Baseline, ● - Acoustic with weight < W_{Middle}, ▲ - Acoustic with weight < $1.1W_0$, ◆ - Weight with SPL < -15 dB, □ - Weight with SPL < -20 dB, O - Compound $\mu_s/\mu_w=2$, Δ - Compound $\mu_s/\mu_w=20$, ◇ - Unoptimized.

Considering Fig. 4 for the circumferential variation, at the lighter weights, the interior noise level difference between the optimized and non-optimized models is on the order of 15 dB. Figure 5 indicates that there is greater scatter in the longitudinal results, and that at the lighter weights the noise level difference is on the order of 10 dB. Both figures indicate that the difference increases with increasing weight.

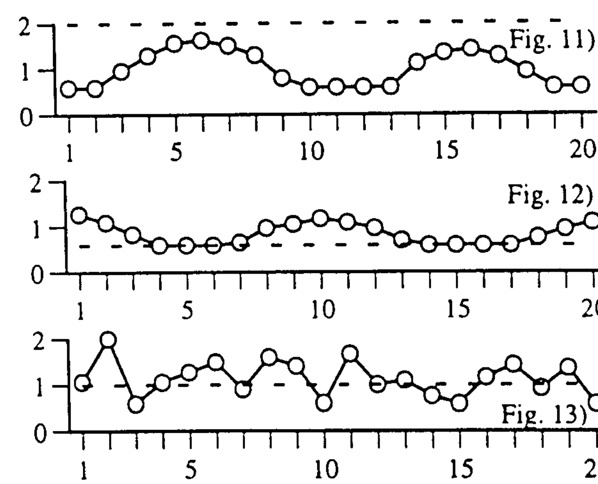
Figures 6 through 10 present the ending design state for a number of the circumferential model analyses. Recall that the design state (thicknesses) has been normalized by the thickness of the "Middle" starting point (1.7 mm). The figure numbers correspond to the numbered data points on Fig. 4. Recall that this model has its design variables grouped circumferentially, such that all elements in a circumferential band have the same thickness. Therefore, the thickness may vary only longitudinally in this model (see Fig. 2). The dashed line in each of the figures indicates the initial starting point. The 20 analyses conducted on this model yielded several alternative trends in the optimal design states. Briefly stated, these alternative trends were for increasing thickness toward the plane of the source, decreasing thickness toward the plane of the source, and a more complex trend of decreasing thickness toward an intermediate point followed by increasing thickness toward the plane of the source.



Figures 6 – 10: Normalized starting and ending design states vs. design variable index, unstiffened circumferential model

The design state in Fig. 6 illustrates the decreasing shell thickness trend. This state was produced by the Baseline formulation. Fig. 7 is a variant of this trend, produced by the Weight objective with a -20 dB acoustic constraint. The design state in Fig. 8 illustrates the trend for increasing thickness toward the plane of the source, and was produced by the Acoustic formulation with a maximum weight constraint of $1.0W_{Middle}$. Fig. 9 illustrates a variant of this trend, produced by the Weight objective with a -20 dB acoustic constraint. Figure 10 illustrates the decreasing/increasing trend. This state was produced by the Acoustic objective with a constraint on the weight of $1.1W_{Middle}$.

Figures 11 through 13 present the ending design state for a number of the longitudinal model analyses. The figure numbers correspond to the numbered data points on Fig. 5. Recall that this model has its design variables grouped longitudinally, such that all elements in a longitudinal band have the same thickness. Therefore, the thickness may vary only circumferentially in this model (see Fig. 3). The dashed line in each of the figures indicates the initial starting point. The 20 analyses conducted on this model yielded several alternative trends in the optimal design states. Briefly stated, these trends were for a periodic variation in thickness around the circumference with maxima at 45° and 135°, a periodic variation in thickness around the circumference with maxima at 0°, 90° and 180°, and a state that at best could be termed chaotic. The design state in Fig. 11 illustrates the first periodic trend. This state was generated by the Acoustic formulation with a constraint on weight of $1.0W_{Middle}$. The design state in Fig. 12 illustrates the second periodic trend, and was also generated by the Acoustic formulation with a constraint on weight of $1.0W_{Middle}$ (note the different starting states). The design state in Fig. 13 illustrates the chaotic state, of which 6 of the 20 analyses could be classed. This state was produced by the Baseline formulation.



Figures 11 – 13: Normalized starting and ending design states vs. design variable index, unstiffened longitudinal model

Conclusions

The results indicate that there is significant potential to reducing tonal interior noise levels within turbo-prop aircraft. For the excitation and frequency conditions considered here, we found that longitudinal variability to yield somewhat greater impact on interior noise levels than circumferential variability.

We do not necessarily propose that airframers should begin fabricating aircraft with different skin thicknesses. We argue, however, that the results indicate that it is possible to improve the interior noise quality within an aircraft fuselage. Varying the skin thickness and/or stiffener geometry may be one approach, whose feasibility depend upon such factors as the fabrication details and structural complexity of a given aircraft. Further, the analyses presented here were for single excitation frequency. Work is in progress to extend the design tool's capability to include multi-frequency capability.

Acknowledgments

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